THE MUNICIPAL UNIVERSITY OF WICHITA

A PROGRESS REPORT ON JET PUMP RESEARCH

by Friedrich Wagner and Charles J. McCune

Engineering Report No. 085

for the Office of Naval Research
Contract Nonr-201(01)

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SUMMARY

This progress report presents the results of a series of tests on a jet pump using steam as the primary fluid to pump atmospheric air. The purpose of the tests was to investigate the mixing process. General calculations and design values are presented with recommendations for future work.

INTRODUCTION

This progress report presents the results of a series of jet pump mixing tube tests. The jet pump used 98 percent dry saturated steam as the primary or actuating fluid pumping atmospheric air, the secondary fluid. The purpose of these tests was the study of the mixing process of the primary and secondary fluids.

The design and tests of the jet pump were performed under the authority of contract number Nonr-201(01), awarded to the University of Wichita by the Air Branch of the Office of Naval Research.

Typical jet pumps are shown in figures 1 and 2. Figure 2 illustrates the pump used for the described tests. The circular-section convergent-divergent nozzle discharged into a straight, cylindrical mixing tube where an isothermal mixing of the jet and the induced air occurred. The inlet for the secondary air was a converging bell mouth. At the beginning of the entrance the total and static pressure of the air was atmospheric (no velocity). At the end of the diffuser, which was used to increase the overall efficiency, the static pressure was atmospheric. The total pressure however was higher at this point because of the dynamic component. The induction forces of the primary jet acted on the secondary air to reduce the static pressure in the region of the nozzle exit. The differential pressure existing between this region and the region at the beginning of the entrance caused flow of secondary air into the mixing tube.

In the ideal case of a jet pump without friction at the mixing-tube wall and without resistance to the flow at the entrance or exit, the jet of primary fluid spreads at a constant angle. In this case the flow of the primary jet and the secondary air is generally in the direction of the axis of the mixing tube. These flow conditions are similar to those of a free jet. In the actual case with viscosity and throttling, the secondary air will flow back along the walls to cover the volumetric need at the exit of the nozzle. The flow in this case causes a condition of vorticity. This condition is amplified with increased entrance and exit throttling and also by a mixing tube of excessive length.

For each application of a jet pump it is necessary to determine the mixing tube dimensions for the highest possible efficiency. For these tests a sufficiently long mixing tube was selected to investigate the mixing process with respect to the above flow conditions.

There are many theories of jet pumps which give the relation of overall performance to the geometry of the jet pump and the main parameters. These theories provide methods of calculating performance only within certain limits. Because viscosity in the mixing zone as well as other important effects have been neglected these theories do not prove satisfactory for the design of high efficiency ejectors. Many design questions concerning the optimum length of the mixing tube must be answered by experimentation and the investigation of the kinetics of the mixing process in which all effects are considered and evaluated in terms of efficiency and specific energy consumption.

SYMBOLS

D	inches	Diameter of mixing tube.
ΔE	BTU/lb	Enthalpy change of primary fluids.
L	inches	Length of mixing tube.
m		Mass ratio, w_s/w_j .
p_o	psfa	Atmospheric pressure.
p_1	psfa	Total pressure of secondary fluid at nozzle exit.
p_2	psfa	Total pressure of combined fluids at end of diffuser.

ΔP_1	psf	$p_c - p_1$
ΔP_2	psf	$p_2 - p_o$
ΔP_t	psf	Total pressure rise.
P_j	psia	Steam pressure to nozzle.
Q_j	ft^3/sec	Volume flow rate, Primary fluid.
Q_s	ft^3/sec	Volume flow rate, Secondary fluid.
T_j	°F abs.	Steam temperature to nozzle.
w_j	lbs/sec	Mass flow rate, Primary fluid.
w_s	lbs/sec	Mass flow rate, Secondary fluid.
W_1	ft-lb/sec	Work input.
W_o	ft-lb/sec	Work output.
η		Efficiency, W_o/W_1

THEORY

The useful work of the jet pump is the summation of the suction work and the blowing work. Useful work:

$$W_o = Q_s(\Delta P_1 + \Delta P_2) + Q_j \Delta P_2$$

where

$$\Delta P_1 = p_o - p_1$$

and

$$\Delta P_2 = p_2 - p_o$$

The required power, W_1, is the product of the energy difference of the primary fluid expanded from the pressure P_j to p_o and the weight of primary fluid flowing per unit of time.

$$\text{Efficiency } \eta = \frac{W_1}{W_o} = \frac{Q_s(\Delta P_1 + \Delta P_2) + Q_j \Delta P_2}{778 \Delta E w_j}$$

APPARATUS

The steam jet pump, designed and fabricated in its most basic form, consisted of the mixing tube, supersonic nozzle, large radius entrance, and small angle diffuser. The entire pump with instrumentation, steam supply line and control valve, was housed in a 30x7½-foot closed trailer. The steam generator was installed in a laboratory building adjacent to the trailer. Figures 3 through 6 illustrate the general layout of the pump and auxiliary equipment.

The jet pump was designed for long-run tests during which detailed pressure readings could be taken. A relatively large pump was designed to accomplish exact measuring of the quantities desired and also because of the high driving energy available. The configuration of the pump was variable in only one respect. By sliding the mixing tube forward on rails over the nozzle, the length to diameter ratio (L/D) could be varied; L being the length of the mixing tube downstream from the exit plane of the nozzle, and D representing the diameter of the mixing tube. The design dimensions of the jet pump components were dictated by experience gained in jet pump work performed in the United States and in Germany. *

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* 1) Rastrelli, Leonard and Snyder, Melvin H.: A Progress Report on Jet-Pump Research Conducted at the University of Wichita. University of Wichita Engineering Report No. 049, August 1951.

2) Wagner, Friedrich: A Contribution to the Development of Jet Pumps. Central Air Documents Office A.T.I. No. 20255, April 1949.

The supersonic convergent-divergent stainless steel nozzle was fabricated with extreme dimensional control and with the interior surface polished to a mirror finish. The nozzle was calculated to discharge steam at a rate of 0.222 lbs. per second for nearly isentropic expansion from an initial condition of 400 psia, dry saturated. The jet velocity from expansion to atmospheric pressure was 2190 ft/sec. While nozzle efficiency tests were not conducted for the initial jet pump tests, the efficiency was assumed to be 96 to 98 percent. This assumption was based on previous fabrication and testing experience.

The physical dimensions of the mixing tube were determined by calculations of the test jet pump without entrance or exit throttling. The mixing tube internal diameter of eight inches was determined from a mass ratio of m = 29 and a total pressure differential between entrance and exit of ΔP_t = 40 psf. The length of the constant cross section tube was chosen as 90 inches. This was a mixing tube length ratio of L/D = 11.25. In earlier investigations the optimum overall efficiency was obtained at a ratio of L/D = 6.4. Since some authors have quoted values of from 4 to 11, the selection of the mixing tube length stated above was felt sufficient to accomplish the tests over a wide range of L/D ratios. Standard steel pipe was used for the mixing tube. The internal surface was honed to a bright finish. The tube was not insulated for the initial tests performed, however insulation will be added for the future tests with superheated steam.

Sixteen No. 70 (.028 in. dia.) holes were drilled through the mixing tube wall for static pressure taps. These holes were approximately equally spaced along the length of the mixing tube. Total pressure readings were taken with a total pressure tube (fig. 8) inserted through additional holes in the mixing tube wall. These holes (24 total) were provided at six stations along the tube and on the cross section centerlines at each station. The total pressure tube was made from .070 inch diameter brass tubing and was housed in a 5/16 x 1/8 inch elliptical steel tube. This steel tube was arranged as a slip fit in a holder in the above mentioned holes. The sides of the holder were aligned with deep longitudinal scribe lines on the mixing tube to assure the proper direction of the total pressure tube in relation to the axis of the mixing tube. Horizontal lines scribed on the elliptical tube indicated the position of the total pressure tube between the mixing tube wall and centerline. With the probe inserted in one of the 24 holes, the remaining 23 holes were plugged. These plugs were machined on the end to fit the inside contour of the mixing tube.

The entrance piece provided a smooth, rounded surface over which the pumped or secondary air could flow into the mixing tube. This entrance was made of wood and was varnished and polished. When installed on the mixing tube it was carefully faired to the inside diameter. Two static pressure openings (.028 in. dia.) were provided in the entrance piece.

A diffuser was used on the end of the mixing tube to accomplish a gradual expansion of the mixed fluids to atmospheric pressure. This diffuser was 40 inches long with an included angle of six degrees. Four static pressure openings (.028 in. dia.) were provided in the diffuser.

Steam as the primary fluid was produced by a Bessler generator, model 85. This generator is a package unit with boiler and auxiliaries incorporated within a single framework. The auxiliaries, feed water pump, blower, and air compressor are driven by "V" belts from a single $7\frac{1}{2}$ HP constant speed motor. The generator produces dry saturated steam at pressures up to 400 psig and a flow rate to 2500 pounds per hour.

The steam generator modulation control-system uses compressed air as the modulating medium. The feed water, fuel, and combustion air controls are connected mechanically to an air cylinder which is operated by a pressure regulator. The pressure regulator is a Bourdon-tube-actuated device which is responsive to changes in steam pressure occasioned by changes in steam output requirements. Feed water enters the boiler through the spillover heat exchanger and feed check valve. The water and fuel are proportioned so that the boiler will evaporate only 85 to 90 percent of the total water fed. The result is wet steam which enters a cyclone separator where the excess water is removed and trapped in the heat exchanger, approximately dry saturated steam being taken off at the top of the separator. Before leaving the boiler, the steam passes through a temperature responsive thermostat which is designed to protect against overheating or loss of boiler water.

The boiler is equipped with a soot blower using steam directly from the steam outlet header through a shut-off valve. The soot blowers consist of a series of corrosion- and heat-resistant hollow blades inserted between pairs of coils. These blades are slotted and arranged to be rotated by hand thus sweeping all surfaces where soot is likely to collect.

Zeolite treated water is strained and fed to the suction inlet to the feed water pump. This pump is a Bessler designed horizontal triplex pump delivering water to the metering valve at approximately 850 psi. The metering valve and by-pass valve, which is automatically operated, regulate the amount of water that passes to the generating tubes. Steam from the tubes passes to a separator where excess moisture is removed and then goes to a header and discharges through the steam stop valve. Attached to this header are the soot blower valve, safety valve, steam pressure gage and pressure regulator.

Natural gas is supplied to a shut-off valve at approximately one psi. A solenoid valve is in the line between the shut-off valve and the burner. It is controlled by the thermostat and pressure regulator switches. When the boiler goes off by excess pressure the solenoid valve shuts off the gas supply. This valve is also actuated in the same manner by excessive temperature, except the gas valve cannot open again until a re-set plunger is pressed. A butterfly valve in the gas line to the burner is mechanically connected to the combustion control assembly to modulate the fire in response to load changes.

Three phase, 220 volt, 60 cycle power is used by the auxiliary-equipment driving motor. The ignition and control circuits are energized by single phase 115 volt. The 115 volt supply is transformed to a 10,000 volt potential between two electrodes in the combustion chamber. The spark formed in the electrode gap ignites the air-fuel mixture that is fed to the combustion chamber from a coned burner head.

The steam line to the nozzle consisted of 1-1/4 inch pipe insulated with 85 percent magnesia, one inch thick. The steam calorimeter was inserted in an enlarged section of the line in descending steam flow. This enlarged section was approximately six feet upstream from the nozzle. Steam entering the supply line at the generator was assumed to be dry and saturated. Because of line pressure drop and a small amount of condensation in the line the steam quality at the nozzle was determined to be approximately 98.5 percent dry. The calculated steam velocity in the supply line to the nozzle was 28.4 feet per second. This initial velocity was neglected in the nozzle design calculations.

To accomplish the purpose of these tests the quantities requiring determination were steam flow rate and secondary air flow rate. In addition to these flow rates, the mixing pattern of the primary and secondary fluids was determined by static pressure readings along the length of the mixing tube and by the total pressure and temperature profiles at the six survey stations along the mixing tube. All instruments except the steam calorimeter were mounted on a panel (fig. 7). The mounted instruments were 18 static pressure manometers, one total pressure manometer, a Brown temperature recorder and the steam pressure and temperature gages. All manometers registered pressure in inches of water. Also on the panel was a data space for recording the date, run number, probe position, nozzle position, barometric pressure, and wet and dry bulb temperatures. The Brown instrument recorded the temperature of the mixed fluid at the total pressure tube as transmitted by an iron-constantan thermocouple attached to the underside of the total pressure tube.

DISCUSSION

An unstable operation of the jet pump was noted during the first operation of the equipment. To aid in analyzing the effect of the instability, it was thought advisable to determine the pump output in terms of mass flow and mass ratio, secondary fluid to primary fluid. These results could then be compared to predicted results and conclusions made about the effect of the instability on pump output.

A cross section survey of one station of the mixing tube was made with total pressure, temperature, and static pressure being recorded photographically. An attempt was made to record the manometer readings at the mid point of their extreme fluctuation. At a later date a complete survey at all stations of the mixing tube was made. During these surveys the operator recorded the maximum and minimum readings for each position of the probe from which the average reading was obtained. These data were then reduced to mass flow and mass ratio and compared with the original results. The first test results indicated a secondary mass flow of 500.6 pounds of air per minute and a mass ratio, m = 28.8. The results of the more accurate survey were 499.4 pounds of air per minute and mass ratio, m = 27.7. As mentioned above, the pump was designed to a predicted mass ratio of m = 29.

The static pressure distribution along the wall is shown in figure 10. At the entrance to the diffuser is shown an adverse condition of static pressure. The excessive length of the mixing tube permits a thick boundary layer to be built up in this region. The total pressure profiles shown in figure 9 indicate the lack of momentum exchange to the flow at the wall with the resultant reversed flow. In future tests with a steady jet and correct mixing tube length this condition should be eliminated to produce smooth diffusion to atmospheric pressure.

The total pressure profiles with fluctuating operation are similar to those of previous tests during which steady flow conditions were maintained. An exception to this similarity are the profiles at stations 3 and 4.

CONCLUSIONS

The test pump arrangement proved to be an excellent installation for the planned study of the mixing process. However, dependable data were impossible to obtain because of the unsteady primary jet. The cause was analyzed as one or a combination of three conditions: (1) a varying steam supply pressure, (2) steam quality fluctuation, or (3) a metastable expansion of the steam in the nozzle combined with droplet lag. Motion pictures were taken (16 frames per second) to determine the relation between steam supply pressure variation and static pressures along the mixing tube. Plots of the data taken from these pictures revealed a close relation in frequency of variation but not in magnitude. The magnitude of static pressure fluctuations increased from small values at the entrance to a maximum at the middle of the mixing tube. The conclusion was that the variation of steam quality was the predominant cause of jet energy variation. The energy variation and improvement of the steam expansion through the nozzle will be corrected by using superheated steam of 400 psia and 200° F. superheat. Except for the additional equipment necessary to provide the superheated steam the future test program can be carried on with very minor changes to the test equipment as now installed.

It is intended to continue the tests as originally planned. Some short test runs will be made to check these reported results. Then a series of tests with throttling at entrance and exit will be run to produce a family of performance curves. Following this will be a systematic variation of the mixing tube length, velocity ratios by changing the cross section areas, and the entrance and diffuser configurations to arrive at the highest overall jet-pump efficiency.

To expand and modify the existing theories of mixing to give practical design information, it will be necessary to obtain the flow parameters velocity, pressure, length and mass ratios. The above outlined tests will provide the necessary data. For comparison of the jet pump with other flow engines and for the development of jet pumps for particular jobs the pump will be evaluated by the efficiency only.

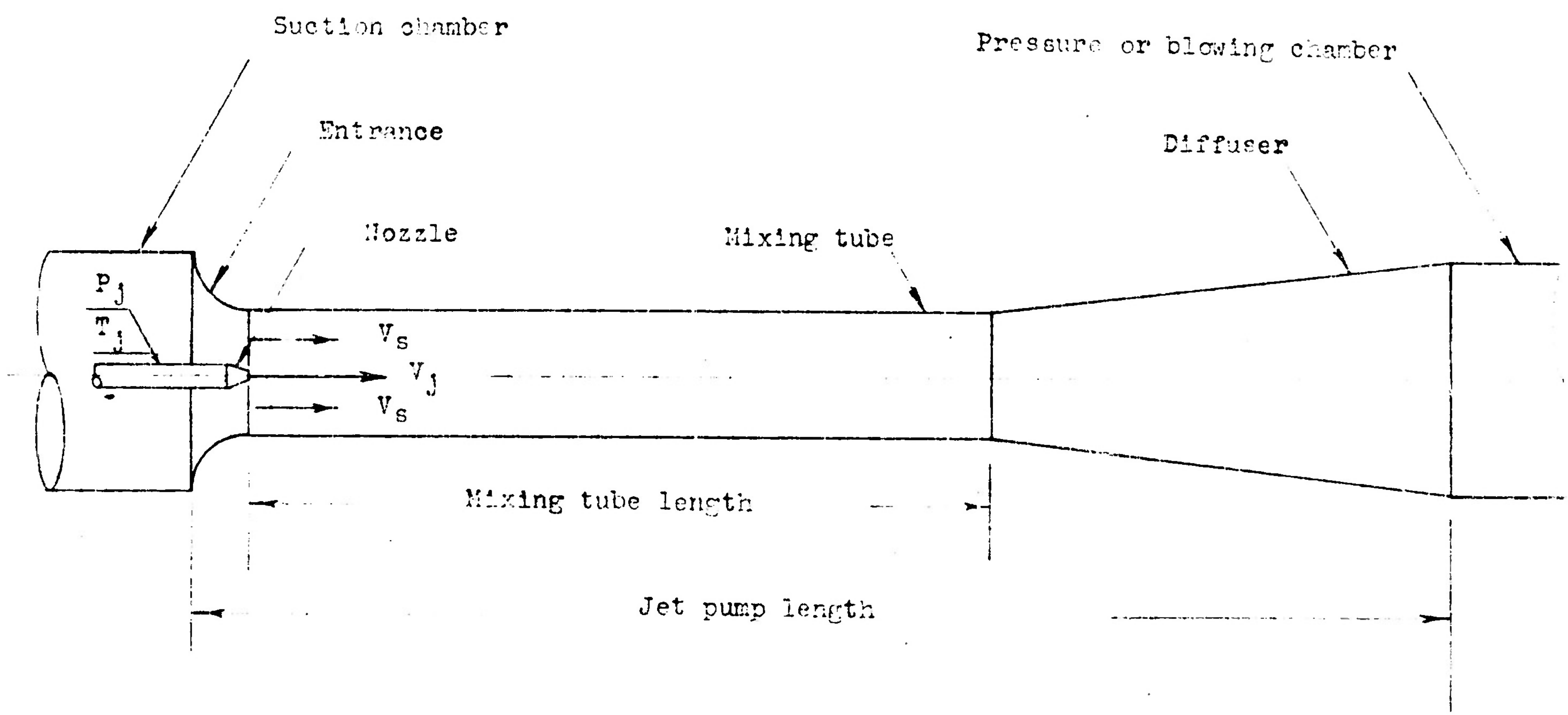

Figure 1.- Typical jet pump with suction and blowing chambers

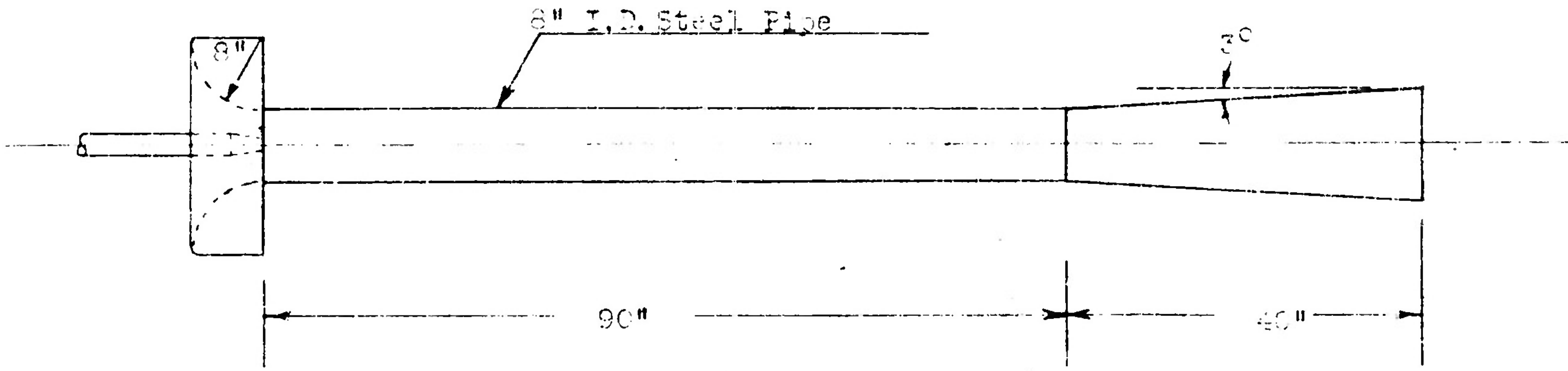

Fig. 2.- Typical jet pump without suction and blowing chambers.

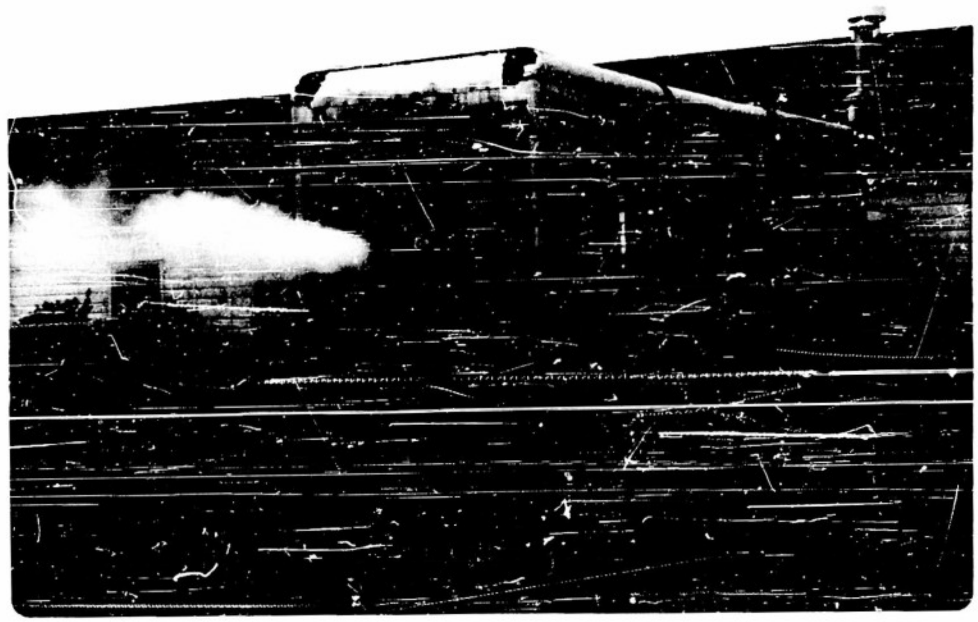

Figure 3.- Portable jet pump laboratory.

Figure 4.- Jet pump installed in trailer.

TEST: Jet Pump
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BY: Wagner and McCune

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DATE: October 1952

Figure 5.- Steam generator.

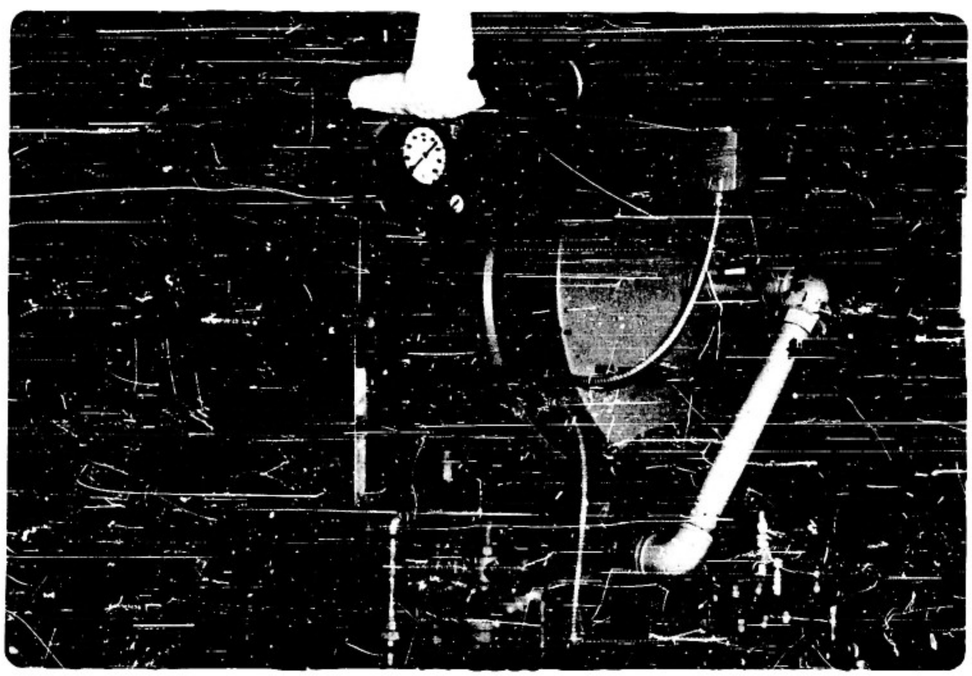

Figure 6.- Steam generator.

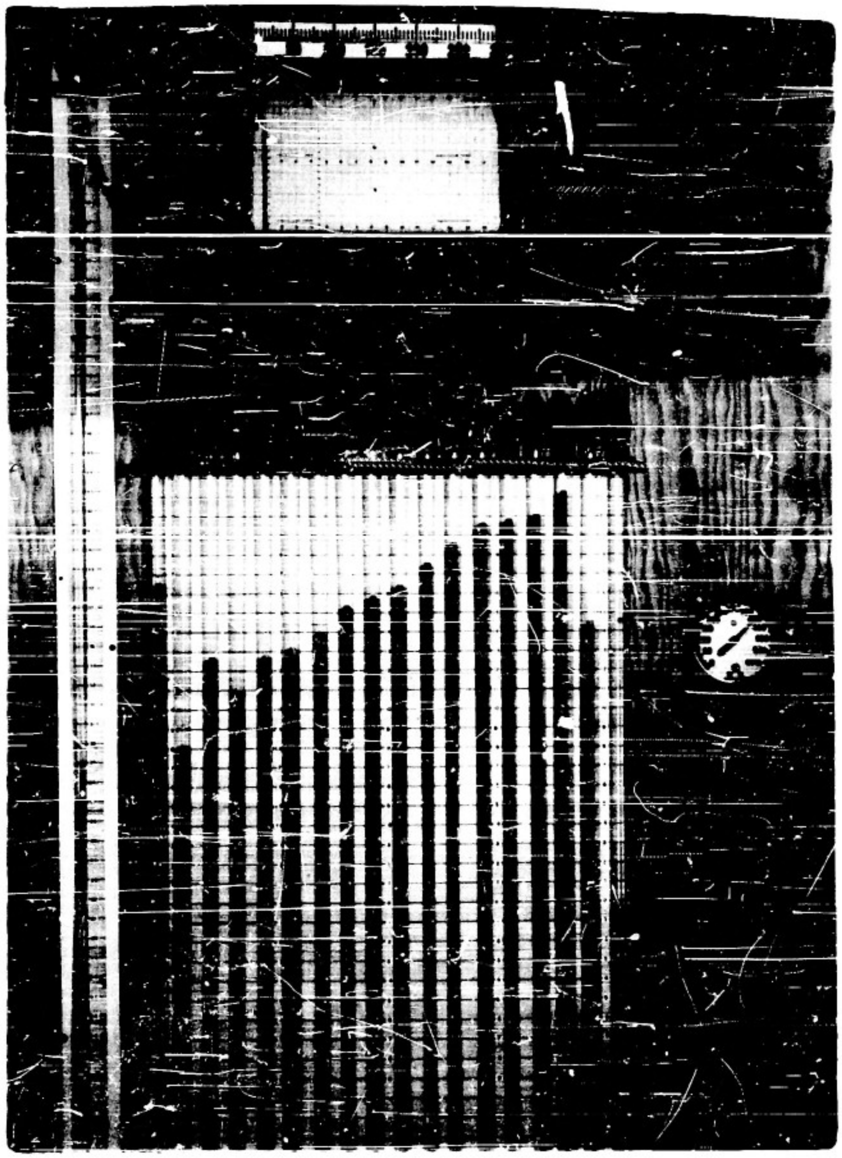

Figure 7.- Instrument panel.

Figure 8.- Total pressure survey instrument.

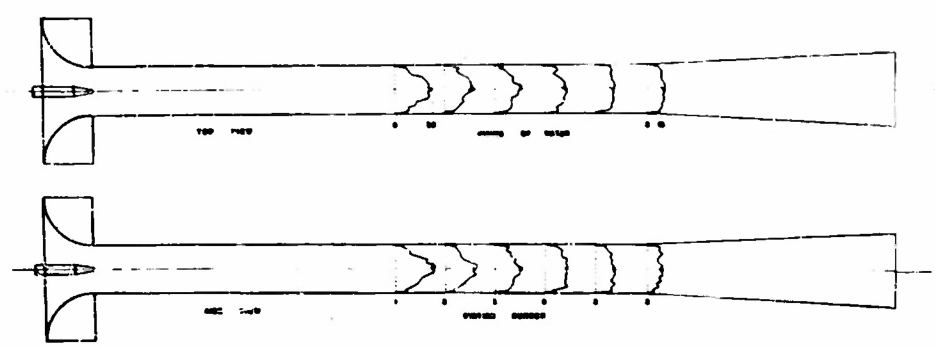

Figure 9.- Total pressure - radial profiles.

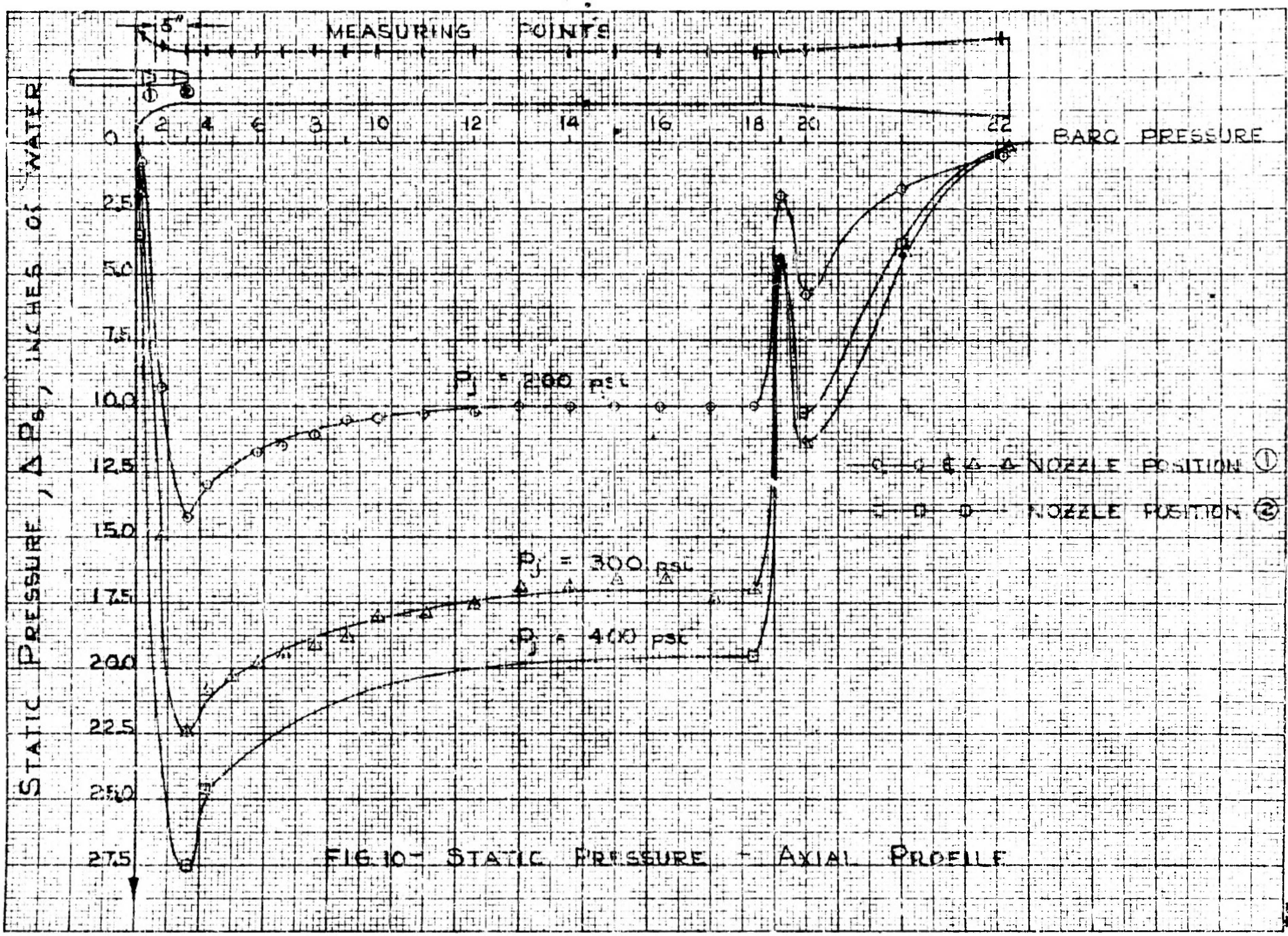

FIG. 10 – STATIC PRESSURE – AXIAL PROFILE

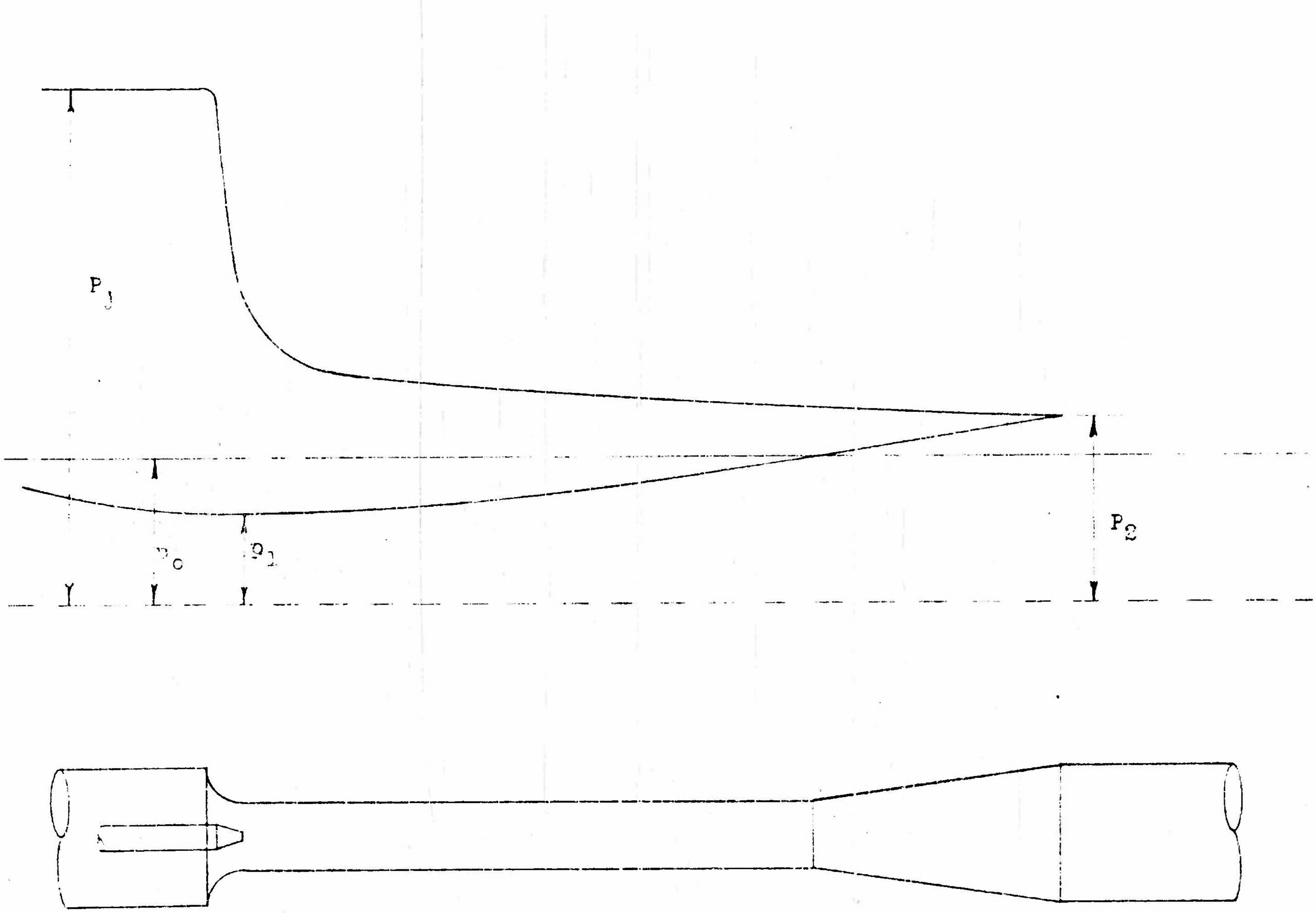

Figure 11.- Total pressure - axial profile, primary and secondary fluids.

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